Prediction Model for Cooling of an Electrical Unit with Time-Dependent Heat Generation

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ABSTRACT: The satisfactory performance of electrical equipments depends on their operating temperature. In order to maintain these devices within the safe temperature limits, an effective cooling is needed. High heat transfer rate, of compact in size and reliable operation are the challenges of a thermal design engineer of electronic equipment. Then, it has been simulated the transient a three-dimensional model to study the heating phenomenon with two assumption values of heat generation. To control for the working of this equipment, cooling process was modelled by choosing one from different cooling technique. Constant low speed fan at one direction of air flow was used for cooling to predict the reducing of heating temperature through working of this equipment.

Numerical Solution of finite difference time domain method (FDTD) has been utilized to simulate the temporal and spatial temperature profiles through two processes, which would minimize the solution errors.

Keywords: cooling by force convection, cooling electronic equipment, thermal analysis of electronic equipment, numerical modeling of cooling electronic equipment.

NOMENCLATURE

Bi: Biot number
g: Acceleration of gravity [m/s^2]
h: Convection heat transfer coefficient [$W/m^2\ ^oC$]
i,j,k: Index increases along x, y, z axis
k: Thermal conductivity [$W/m.^oC$]
L: Number of mesh points in the direction of z-axis
M: Number of mesh points in the direction of x-axis
n: Integer
N: Number of mesh points in the direction of y-axis
t: Time [Second]
T: Temperature [^oC]
u: Velocity in x-direction [m/s]
x, y, z: Distance [m]
c_p: Specific heat at constant pressure [$J/kg.^oC$]
q: Heat generation [W/m^3]
T_∞: Ambient temperature [^oC]
α: Thermal diffusivity [m^2/s]
ρ: Density [kg/m^3]
λ: Convergence factor

1. INTRODUCTION

The heat mechanism of heat generation from the electrical equipment through its working and the air around is often described by means of thermal model. Thus, temperature profiles as a function of time and at various locations in the bulk of the material were calculated.

Most of the theoretical work has centered on the solution of the momentum and energy equations to show how the air around was influenced by heat dissipation from this equipment. While, Eriksson and Sunden [1] developed a numerical model based on finite volume method with implicit technique for estimating the transient temperature response and cooling rate of an electrical unit with time dependent internal heat generation. The accuracy and reliability of the model were established by considering experimental case studies. The electrical unit is placed in a closed box with cooling channels on two opposite sides. Measurements of the surface temperatures and cooling rate with time are achieved. Increasing the flow rate has a minor effect on the temperature. When water is used as coolant the cooling rate is considerably higher than when oil is used as coolant. Natural convection cooling, in particular, results simple, cheap and reliable. In compact, low power consumption electronic equipments, in sealed system or available for the cooling technique of the system. It is also the only way to avoid or

reduce the acoustic noise related with fan or liquid pumps characterizing other kinds of cooling systems. More complex cooling technique is necessary only in high power applications. In these cases we have to face with increase of costs and added cooling related space [2]

The present work predicts two models to show how the heat transfer mechanism was happened between a closed casing and air around, which was assumed as electronic equipment. This closed box was heated by heater, which was fixed at the bottom of this box. This position of heater was made as heat source per unit volume of a rectangular closed box. First model predicts the temperatures distribution over all the surfaces of a closed casing. These risings of temperatures were produced by heating with a heat source (heater) at two different values of power 25W and 70W respectively. Two approaches were assumed firstly, the effect of natural air around of these heating temperatures as natural convection.

In second model, it has been fixed a fan in cabinet in order to exhaust the air around, and then the heat mechanism was assumed between heating temperature and around air moving by fan (i.e. forced convection).The influence of the thermal properties of box material were taken in two models.

2. MATHEMATICAL ANALYSIS

2.1 Heating Model

The general three-dimensional of heat conduction equation is given by reference [3]. This has been used to describe the temperature distribution within quarter rectangular box as shown in (Fig.1).

$$\rho c \frac{\partial T}{\partial t} = \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right) + \dot{q}(x,y,z,t) \qquad (1)$$

Assuming that the thermal properties are independent of temperature, the solid is taken to be homogeneous and isotropic, equation (1) reduced to

$$\frac{1}{\alpha}\frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{k} \qquad (2)$$

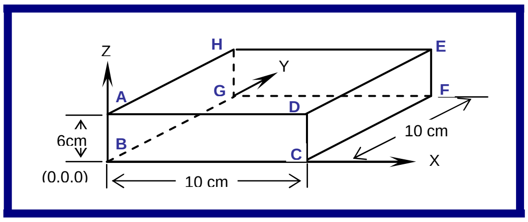


Fig.1. the Cartesian coordinate system.

2.1.1 Numerical Analysis of Heat Transfer Model

The principle of conservation of energy with the control-volume methods were used to make the analysis of temperature distribution through surface of different planes of rectangular box. Consider a rectangular plate in which the temperature distribution at front planes of quarter rectangular box. Let the physical properties be constant and uniform and the boundary condition be as shown in (Fig. 2):

At x=L , $$-k\frac{\partial T}{\partial x} = h(T - T_\infty) \qquad (3)$$

z=0 , $$\frac{\partial T}{\partial z} = 0 \qquad (4)$$

(through line BC)

z=N , $$-k\frac{\partial T}{\partial z} = h(T - T_\infty) \qquad (5)$$

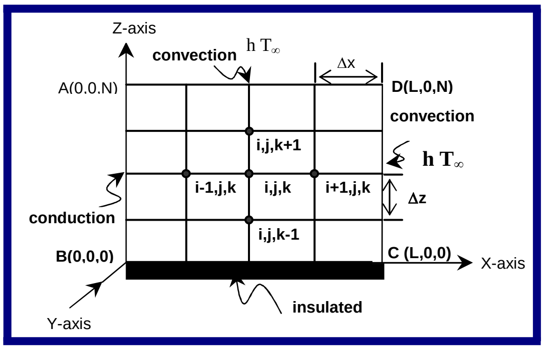


Fig.2. grid layout and boundary conditions for (x-z) plane (front plane).

To handle the problem numerically, it has been divided the width BC into (L-1) equal parts of length Δx, and the length CD into (N-1) equal parts of length Δz. The resulting grid will have L times N nodal points. All the points laying on the boundary A – B – C – D – A are called boundary nodes, and the rest of the points of the grid are called internal nodes. It is convenient to refer to a nodal point by its nodal coordinates (i,j,k) and the physical coordinates (x,y,z) is $(i-1)\,\Delta x = x_i$, $(j-1)\,\Delta y = y_j$, and $(k-1)\,\Delta z = z_k$.

At point B as shown in (Fig. 2) x=y=0, and z=0, upon the boundary conditions were assumed. Then $T^n_{i+1,j,k} = T^n_{i-1,j,k}$ and $T^n_{i,j+1,k} = T^n_{i,j-1,k}$ symmetrical about this point. Therefore, $\frac{\partial^2 T}{\partial x^2} = \frac{\partial^2 T}{\partial y^2}$, and $\frac{\partial T}{\partial z} = 0$ then equation (2) becomes:

$$T^{n+1}_{i,j,k} = 4\lambda T^n_{i+1,j,k} + (1-6\lambda)T^n_{i,j,k} + 2\lambda T^n_{i,j,k+1} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \qquad (6)$$

As a computational grid of the three dimensional model, see (Fig. 2).

Equation (6) is referred to as the equation for node (i,j,k), since it is obtained by performing an energy balance on that element. A complete set of equations will be obtained by writing an energy balance for the boundary nodes. Special care must be exercised when handling corner nodes at A , C and D.

Through edge BC, that edge is insulated and subjected the boundary conditions z=0, when $\frac{\partial T}{\partial z}=0$. Then energy balance was employed to calculate the temperature distribution through edge BC. Hence, equation (2) becomes as:

$$T^{n+1}_{i,j,k} = \lambda T^{n}_{i+1,j,k} + (1-6\lambda)\, T^{n}_{i,j,k} + 2\lambda T^{n}_{i,j,k+1} + \lambda T^{n}_{i-1,j,k} + 2\lambda T^{n}_{i,j+1,k} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (7)$$

Through edge BG, where x=0 and z=0, when $\frac{\partial T}{\partial z}=0$. The new form of equation (2) becomes:

$$T^{n+1}_{i,j,k} = 2\lambda T^{n}_{i+1,j,k} + (1-6\lambda)\, T^{n}_{i,j,k} + 2\lambda T^{n}_{i,j,k+1} + \lambda T^{n}_{i,j+1,k} + \lambda T^{n}_{i,j-1,k} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (8)$$

Through edge AB, where x=y=0 allows calculating the temperature profile against z-axis, the form of equation (2) becomes:

$$T^{n+1}_{i,j,k} = 4\lambda T^{n}_{i+1,j,k} + (1-6\lambda)\, T^{n}_{i,j,k} + \lambda T^{n}_{i,j,k+1} + \lambda T^{n}_{i,j,k-1} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (9)$$

The computer program gives reliability for calculation of the temperature profiles of all inside nodes in x-z plane at y=0.

The internal nodes as shown in (Fig. 2), equation (2) becomes:

$$T^{n+1}_{i,j,k} = \lambda T^{n}_{i+1,j,k} + (1-6\lambda)\, T^{n}_{i,j,k} + \lambda T^{n}_{i-1,j,k} + 2\lambda T^{n}_{i,j+1,k} + \lambda T^{n}_{i,j,k+1} + \lambda T^{n}_{i,j,k-1} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (10)$$

At edge CD boundary condition (3) was applied (i.e. conduction=convection)

$$-k \left.\frac{\partial T}{\partial x}\right|_{x=L} = h\left(T - T_{\infty}\right) \qquad (11)$$

Where

$\frac{h\,\Delta x}{k}$ = Biot number = Bi

When applying energy balance on this edge CD, then equation (2) becomes:

$$T^{n+1}_{i,j,k} = 2\lambda\, T^{n}_{i-1,j,k} + (1-6\lambda-2\lambda Bi)\, T^{n}_{i,j,k} + \lambda T^{n}_{i,j,k-1} + \lambda T^{n}_{i,j,k+1} + 2\lambda T^{n}_{i,j+1,k} + 2\lambda Bi\, T_{\infty} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (12)$$

At edge AD when applying the boundary conditions, Then equation (2) is represented as:

$$T^{n+1}_{i,j,k} = \lambda T^{n}_{i+1,j,k} + (1-6\lambda-2\lambda Bi)\, T^{n}_{i,j,k} + \lambda T^{n}_{i-1,j,k} + 2\lambda T^{n}_{i,j,k-1} + 2\lambda T^{n}_{i,j+1,k} + 2\lambda Bi\, T_{\infty} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (13)$$

Corner C, which it insulated at z=0 when $\frac{\partial T}{\partial z}=0$ .Then equation (2) is written as:

$$T^{n+1}_{i,j,k} = 2\lambda T^{n}_{i-1,j,k} + (1-6\lambda-2\lambda Bi)\, T^{n}_{i,j,k} + 2\lambda T^{n}_{i,j,k+1} + 2\lambda T^{n}_{i,j+1,k} + 2\lambda Bi\, T_{\infty} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (14)$$

At Corner D equation (2) represents as subjected the energy balance on the corner D as:

$$T^{n+1}_{i,j,k} = 2\lambda T^{n}_{i-1,j,k} + (1-6\lambda-4\lambda Bi)\, T^{n}_{i,j,k} + 2\lambda T^{n}_{i,j,k-1} + 2\lambda T^{n}_{i,j+1,k} + 4\lambda Bi\, T_{\infty} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (15)$$

At the energy balance for corner A, then equation (2) becomes as:

$$T^{n+1}_{i,j,k} = 2\lambda T^{n}_{i+1,j,k} + (1-6\lambda-2\lambda Bi)\, T^{n}_{i,j,k} + 2\lambda T^{n}_{i,j,k-1} + 2\lambda T^{n}_{i,j+1,k} + 2\lambda Bi\, T_{\infty} + \frac{\dot{q}_{(i,j,k)}\, \alpha\, \Delta t}{k} \qquad (16)$$

Analysis in more details for all surface of the rectangular box was described in reference [4].

2.2 Cooling Model

2.2.1 Numerical Analysis of Cooling Model

Energy equation was used in order to show how the temperatures of faces decreasing through cooling process with fan has been assumed the flow of air cooling through x-axis with neglected viscous energy dissipation, pressure gradient, under these conditions energy equation as given in reference [5] becomes:

$$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) + \frac{\dot{q}_{(i,j,k)}\alpha}{k} \qquad (17)$$

Same analysis of conservation of energy with control-volume methods were used to calculate cooling temperature distribution through surface of different planes of rectangular box as shown in (Fig. 1). Same assumptions of boundary conditions were used through heating model as reported in section (2.1).

Applying equation (17) for all the outside and corners of rectangular box, which was effected by the flow air of fan in order to reduce these temperatures from high range to lower range (i.e. cooling by forced convection). This equation was represented by different form depending upon the boundary conditions which was assumed as in previous model (i.e. heating model).

In front plane (x-z) through edge CD as shown in (Fig. 2) equation (19) becomes:

$$T_{i,j,k}^{n+1} = 2\lambda T_{i,j+1,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \lambda T_{i,j,k-1}^{n} + \lambda T_{i,j,k+1}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \quad (18)$$

Also edge AD in the same plane (x-z) as shown in fig 2. After applying the initial and boundary conditions. Equation (17) is represented as:

$$T_{i,j,k}^{n+1} = 2\lambda T_{i,j+1,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \lambda T_{i+1,j,k}^{n} + \lambda T_{i-1,j,k}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \quad (19)$$

Corner C in front of quarter rectangular box and with the same boundary conditions in heating model after applying equation (17), then equation of temperature distribution through corner C becomes:

$$T_{i,j,k}^{n+1} = 2\lambda T_{i,j+1,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + 2\lambda T_{i,j,k+1}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \quad (20)$$

Corner D in front plane (x-z), equation (17) is illustrated as:

$$T_{i,j,k}^{n+1} = 2\lambda T_{i,j+1,k}^{n} + \left(1-4\lambda+2\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \left(2\lambda-2\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\,\alpha\,\Delta t}{k} \quad (21)$$

Corner A with feeding the boundary conditions, then equation (17) becomes:

$$T_{i,j,k}^{n+1} = 2\lambda T_{i,j+1,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + 2\lambda T_{i+1,j,k}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \quad (22)$$

In the right plane (y-z) all the insides nodes obeyed the same boundary conditions as discussed, and there is a flow of air in normal direction to the plane, thus equation (17) becomes:

$$T_{i,j,k}^{n+1} = \lambda T_{i,j+1,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \lambda T_{i,j,k-1}^{n} + \lambda T_{i,j,k+1}^{n} + \lambda T_{i,j-1,k}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\,\alpha\,\Delta t}{k} \quad (23)$$

For edge CF in right plane, equation (17) becomes:

$$T_{i,j,k}^{n+1} = \lambda T_{i,j+1,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + 2\lambda T_{i,j,k+1}^{n} + \lambda T_{i,j-1,k}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \quad (24)$$

Through edge EF the temperature distribution through cooling with the same boundary conditions of heating model and with effected by the air of fan (force convection), equation (17) becomes as:

$$T_{i,j,k}^{n+1} = \lambda T_{i,j,k-1}^{n} + \left(1-4\lambda+2\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \lambda T_{i,j,k+1}^{n} + \left(2\lambda-2\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\,\alpha\,\Delta t}{k} \quad (25)$$

At edge DE in right plane (y-z), equation (17) becomes:

$$T_{i,j,k}^{n+1} = \lambda T_{i,j+1,k}^{n} + \left(1-4\lambda+2\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \lambda T_{i,j-1,k}^{n} + \left(2\lambda-2\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\,\alpha\,\Delta t}{k} \quad (26)$$

Corner F in right plane (y-z) with the previous boundary conditions and with forced air cooling, then equation (17) becomes:

$$T_{i,j,k}^{n+1} = 2\lambda T_{i,j,k+1}^{n} + \left(1-4\lambda+2\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \left(2\lambda-2\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \quad (27)$$

At corner E , the new form of equation (17) becomes as:

$$T_{i,j,k}^{n+1} = \left(1-3\lambda+3\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \left(3\lambda-3\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\,\alpha\,\Delta t}{k} \quad (28)$$

For rear front plane (x-z), same boundary conditions and same analysis as right side plane (y-z) plane was made.

For top plane (x-y). For all insides nodes in top plane, with the same boundary conditions and with air flow in x-direction by fan. Then, equation (17) represents as:

$$T_{i,j,k}^{n+1} = \lambda T_{i+1,j,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \lambda T_{i-1,j,k}^{n} + \lambda T_{i,j+1,k}^{n} + \lambda T_{i,j-1,k}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\,\alpha\,\Delta t}{k} \quad (29)$$

At edge AH then the new form of equation (17) becomes as:

$$T_{i,j,k}^{n+1} = 2\lambda T_{i+1,j,k}^{n} + \left(1-5\lambda+\lambda Bi+\frac{u\Delta t Bi}{\Delta x}\right)T_{i,j,k}^{n} + \lambda T_{i,j+1,k}^{n} + \lambda T_{i,j-1,k}^{n} + \left(\lambda-\lambda Bi-\frac{u\Delta t Bi}{\Delta x}\right)T_{\infty} + \frac{\dot{q}_{(i,j,k)}\alpha\,\Delta t}{k} \quad (30)$$

3. RESULTS AND DISCUSSIONS

(Fig. 3) shows the temporal variation of temperature in two corners C and D for both

heating and cooling model with heat generation 29166.666 W/m^3. The solid line represents the prediction temperature variation with time through heating process, while the dash line illustrates the temperature profiles of cooling process. It has been shown the cooling model shows the reduction of temperature values due to the effect of forced convection by fan. It has been indicated that the temperature values increasing with time intervals. It point that the temperature value in corner C is greater than the temperature value of corner D. This is due to the position of corner C at the bottom of brass box, and it is near of the heat source. The reduction values of temperature from heating to cooling in (Fig. 3) in two corners were about 20^{o}C at 900 second. This reduction decreasing with time, this value becomes about 11^{o}C at 3600s. Also, in these figures the value of heat generation 29166.666W/m^3 gives a high temporal variation of temperature of these two corners than the value of heat generation 10416.666W/m^3.

(Fig. 4) and (Fig. 5) show the spatial temperature distribution of edge CD, which represents z-axis in right side plane (y-z) at every 900 seconds time intervals, and with a two values of heat generation for heating and cooling process. It has been shown more gradient temperature at the surface until 2cm after that the temperature gradient returning to a shallow negative gradient. It has been looking as a steady linear response after distance of 2cm with different time intervals. The reduction of thermal analysis (cooling model) represents a dash line. The same response and behaviour of heating process has been taken. More negative gradient was seen at time 3600 seconds in (Fig. 4) than (Fig. 5). This due to the value of heat generation.

(Fig. 6) and (Fig. 7) illustrate the isothermal contour map of back front plane of brass box (x-z plane) for heating and cooling process. It has been pointed that temperature is very high at the center node of this plane and starting to decrease when move faraway from the center. Same response was given from the isothermal contour of cooling as shown in (Fig. 7). These two figures were predicted at time of 3600 seconds, and the heat source value 29166.666 W/m^3.

(Fig. 8) designates the spatial temperature distributions through the edge AD in the top plane for heating and cooling simulation at different time intervals, and heat source value of 29166.666 W/m^3. Solid line represents the prediction results of heating model, while the dash line represents of cooling model. This figure was indicated higher heating temperature profiles for this edge than other edges of the top plane. It has been indicated that the maximum temperature was at the center of top front plane. That was consistent with the reference [6].

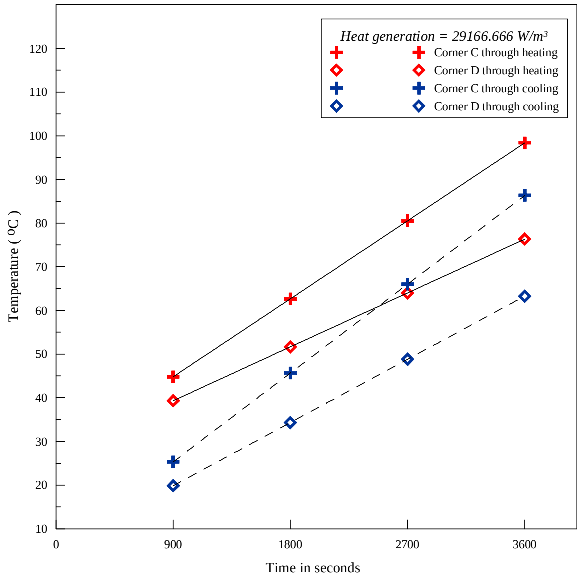


Fig.3. temporal variation of temperature in corner C & D of brass box through heating and cooling models.

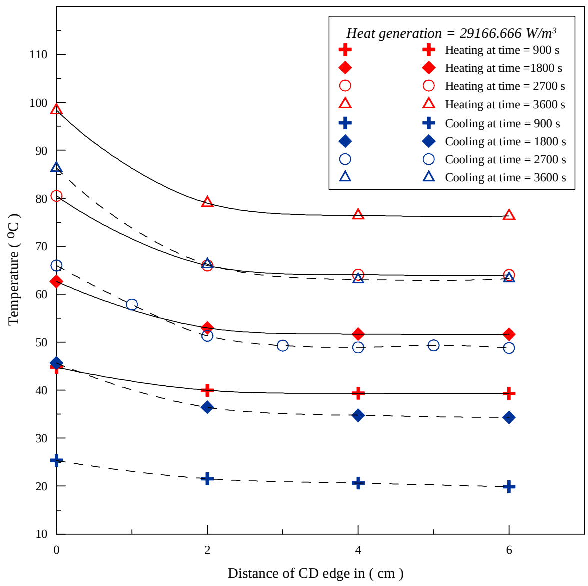


Fig. 4. temperature profile against CD edge for different heating and cooling times in brass box.

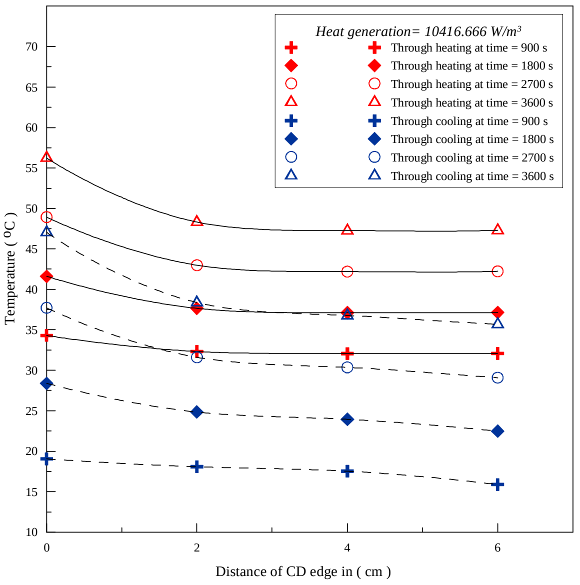


Fig.5. temperature profile against CD edge for different heating and cooling times in brass box.

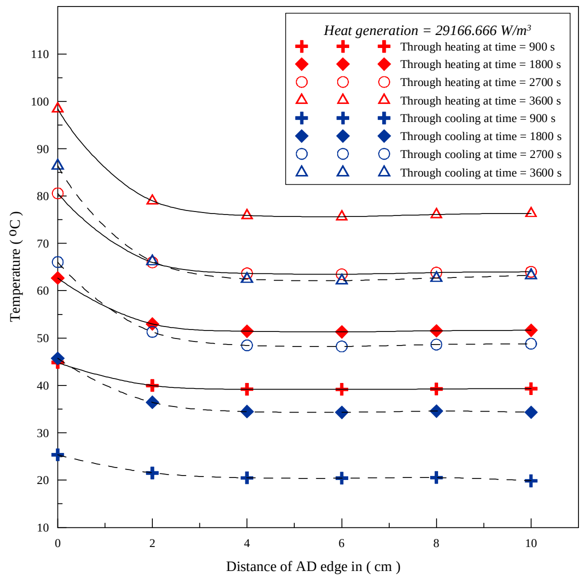


Fig.8. temperature profile against AD edge for different heating and cooling times in brass box.

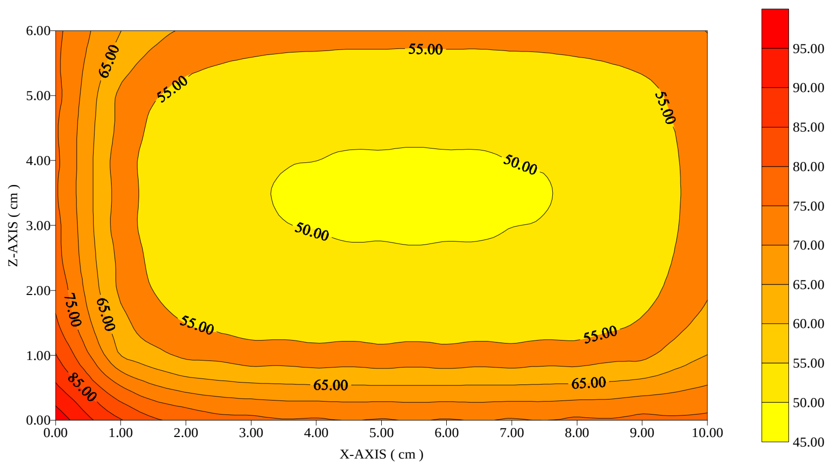


Fig. 6. isothermal contour map through heating for back front plane of brass box with heat generation 29166.666 W/m^3, and time 3600 s.

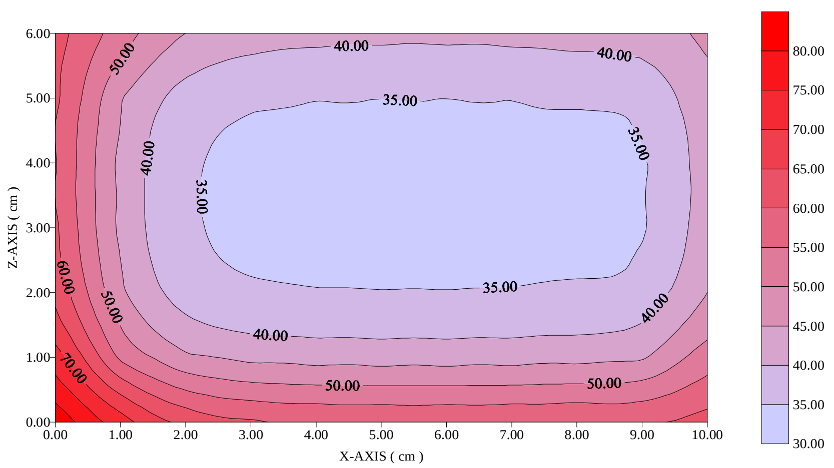


Fig.7. isothermal contour map through cooling for back front plane of brass box with heat generation 29166.666 W/m^3, and time 3600 s.

4. CONCLUSIONS

1. Two models were simulated in this work. First model was predicted to present the spatial and temporal temperatures distribution of rectangular box and all outside corners, and all outside edge planes. The prediction model was unsteady, and in three dimensions with assumption of two values of heat generation in order to show the mechanism of heat transfer through heating process due its work function. While the second model was predicted to present the cooling process. This process was important to avoid this equipment from damage due to the heat generation producing through working. This model in three-dimension, Cartesian coordinate, unsteady state condition. In this model it was assumed at constant air velocity, which was about 0.9195 m/s producing by low power fan. These two models give a better understanding of the heating and cooling process, and give more realistic results, especially for presentation of the spatial and temporal temperature distribution through both processes.
2. Numerical solution using the explicit technique, have been performed to the basic 3-dimensional differential equation of heat diffusion in electrical equipment and energy equation of cooling this equipment, utilizing finite difference analysis for the surface and the distributed heat source of rectangular equipment type.

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